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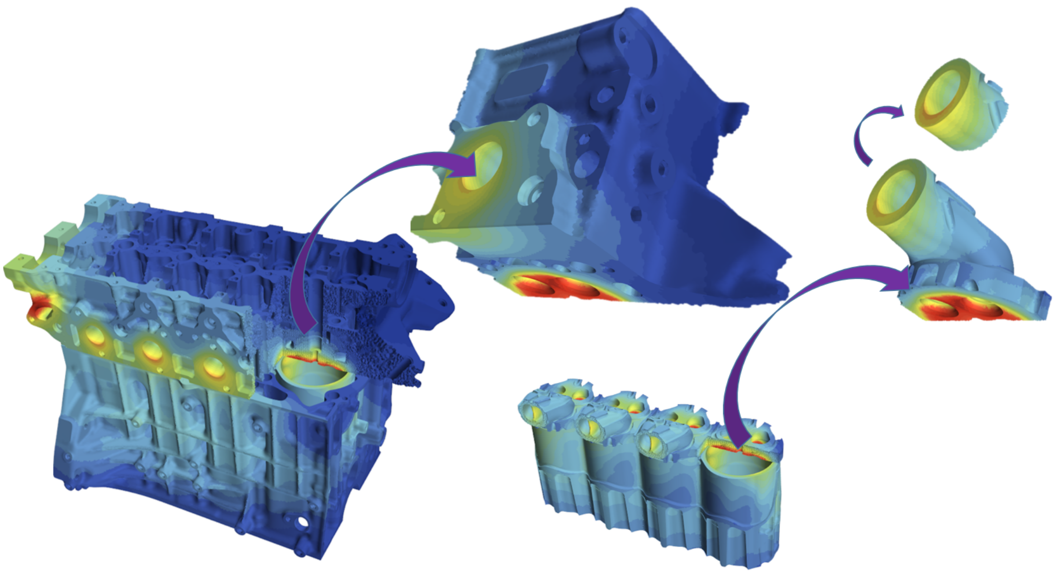

Integration of Thermal and Performance Models - Complete Engine Model

Master's Thesis in Automotive Engineering

Rangakishen Mavanur Sampath
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Department of Combustion
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2017

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Cover: Representation of engine discretization for thermal flow.

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Abstract

The demand for reducing energy expenditure and increasing efficiency have been two of the main driving forces behind the research and development in many major industries today. It is the case even with the automotive industry. The need for higher efficiency propulsion systems is further intensified by the fact that the automotive industry is one of the largest contributor to air pollution. Increasing efficiency allows not only to save energy but also to decrease the emissions per unit of work derived. This push for efficiency has seen development of various solutions such as electric and hybrid powertrains, downsizing of combustion engines while increasing the specific power output etcetera.

While focusing on energy efficiency in any system, it is critical to keep accurate track of the flow of energy in it. Energy in a combustion engine is input into the system as chemical potential energy. This is transformed into heat energy and then mechanical energy. Parts of both heat and mechanical energies are lost in various ways in the system - thermodynamic inefficiency, friction etcetera. Mono dimensional models to predict the flow of heat in an engine, predict the performance of an engine, predict the friction load on the engine, simulate the oil flow in an engine and simulate the cooling system of the engine have all been developed. But they work independently with main boundary conditions being input by the user using experimental data.

In this thesis, all the models mentioned above are integrated to obtain one single model with two way coupling between the models allowing dynamic boundary conditions to be input. This allows more accurate tracking of energy in the system, especially in transient cases like engine start up. The model is configured for three operational modes - steady state full load, steady state part load and transient part load.

The steady state models are run and the results are used to verify energy conservation and to see the deviation from the stand alone models. The transient model is setup for various warm up strategies and study their effect on friction, temperature profiles and fuel consumption. The strategies considered in this thesis are the coupling, decoupling and timed decoupled of the engine oil cooler and variation of the oil quantity. The model was also used to conduct a parametric study of the dependency of friction on oil and coolant temperatures.

Keywords: thermal model, performance model, flow of heat, two way coupling, warm up strategies, steady state model, transient model

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Göteborg 2017-06-06

Nagabhushan and Rangakishen

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Introduction

1.1 Background

Ever increasing number of vehicles has led to implementation of stringent emission laws by the authorities. It is very important for the car manufacturers to adhere to the emission laws while meeting customer demands like fuel consumption, performance and durability. Internal combustion engines are still the preferred choice as prime movers for vehicles. Cars powered by gasoline and diesel engines hold majority of the sales stake[1]. In EU, 53% of all new cars sold in 2014 were powered by diesel engines[1].

At present, it is estimated that passenger cars are responsible for 12% of the total carbon dioxide emission in EU. In 2015, the law demanded that the cars registered in EU should emit less than 130 grams of CO2 per kilometre on average which translates to a fuel consumption of 5.6 litres per 100 km for petrol or 4.9 l/100 km for diesel. By 2020 the law mandates that new cars achieve a fleet average emission of 95 grams of CO2 per kilometre which translates to a fuel consumption of 4.1 l/100 km for petrol or 3.6 l/ 100 km for diesel[1]. From the information it is evident that the future diesel engines should be very efficient to be able to meet the emission regulations while maintaining competitive performance.

An internal combustion engine is a device that converts combustion resultant heat energy into mechanical energy. The efficiency and the power output of the engine depends on the effective utilization of available heat energy. However conversion of the entire generated heat energy to usable mechanical output is thermodynamically not feasible as some of the generated energy is lost into engine components, environment, coolant and lubrication oil. It is very important to study the flow of heat through different parts of the engine and to the environment, one of the effective way to perform this during the development stage of the engine is by using a heat balance model. The model can be a high resolution 3D model or a fast running 1D thermal mass model[2]. A 3D model results in high accuracy but requires more computational time and cost. A 1D model requires less computational time and memory but the accuracy is less compared to the 3D model. GT-SUITE is a commercial simulation software which is widely used in automotive industry for 1D simulations. It is an object based software, that can be used for wide range of modelling applications. A 1D model created in GT-SUITE based on a 3D model is used in this work. Different heat sources and sinks are identified and the necessary boundary conditions are obtained from models as well as from measurements. The objective of the work, methodology, models used, results and conclusions are detailed in the upcoming sections.

1.2 Aim

The aim of the thesis is to establish two way coupling between an engine performance model consisting of combustion, gas exchange, friction, and the lubrication, cooling models with a thermal mass model. In the final integrated model there will be exchange of data between all the models which provides dynamically changing boundary conditions resulting in a more accurate simulation of the engine. Later the integrated model will be calibrated and used for a parametric study of influence of oil and coolant on friction and fuel consumption.

1.3 Objective

- Understanding the engine and thermal models through literature survey
- Study of the simulation platform GT-SUITE and methods of coupling the models
- Integration of gas exchange, cooling and oil models with the thermal mass model
- Calibration of model using the test results from test rig data
- Optimization of the model to simulate thermal behaviour during steady state and transient operations
- Study on the effect of oil and coolant circuits on fuel consumption during engine warm up

1.4 Limitations

This work focuses on the thermal behaviour and performance of only the engine and not those of the complete vehicle. The thermal model of the engine considered is a fast running 1D model which is coarsely discretized. The geometric properties of the components in the thermal and the gas exchange model had notable differences. To eliminate the effects of the geometrical difference the properties in the thermal model were considered in the integrated model.

The friction data was available for a steady state part load condition, based on this the friction for other operating conditions have been approximately modelled.

The thermal mass model used to simulate transient operating condition is not calibrated for full load operating case thus the integration is done only for part load operating condition. The Integrated engine model is not calibrated for the full torque operating conditions.

2

Theory

2.1 Internal Combustion Engines

Internal combustion engines are energy converters which convert chemical potential energy into heat and mechanical energy. The conversion follows the thermodynamic principles. Thermodynamic cycles like the Otto Cycle, Diesel cycle etcetera follow different principles and differ in their overall efficiency. Modern diesel engines have thermal efficiency of about 30-40%,[2] which means that only part of the chemical energy input is available as usable mechanical shaft power. The rest of the energy is lost in the form of heat energy to heat up the engine mass, oil mass, transported to environment through coolant and friction in the engine as shown in the figure 2.1.

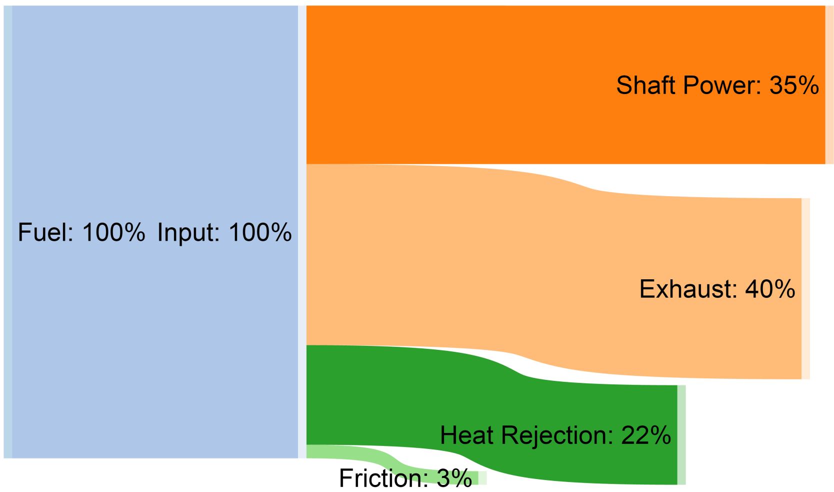

Figure 2.1: Sankey diagram showing the distribution of energy

Various technologies are being used to recapture the heat energy lost in an effort to increase the usable work output from the engine. Turbocharging is one such technology which utilizes some of the energy in the exhaust to increase the mechanical output.

In an effort to reduce emissions, some compromises have to be made which decreases the combustion efficiency of the engine. Exhaust Gas Recirculation, for example, reduces the efficiency of the engine, but is necessary to reduce the gas temperature which reduces NOx emissions.

Complex exhaust after-treatment systems have been developed to reduce the tail pipe emissions. These systems depend critically on the temperature of the exhaust gases. Thus it is very important to maintain the exhaust gas temperatures within a given range.

Given these factors, it is important to have detailed knowledge of the transport of heat from the combustion chamber to various parts of the engine. In this work the heat flow in a Diesel engine developed by Volvo Car corporation will be studied in steady state and transient operating conditions such as warm up.

2.2 Engine Gas Exchange Model

The engine in focus is a 4 cylinder diesel engine developed and produced by Volvo Car corporation, apart from powering the vehicle the engine acts as a source of heat for various appliances. During warm-up phase the availability of heat is scarce which significantly influence emission and fuel consumption to a large extent. It is important that the engine operates efficiently while satisfying all the requirements and to achieve this proper study of the engine and its processes from the development stage is necessary. The working cycles of a diesel engine occurs in fraction of seconds and it is difficult to simulate the transient processes accurately enough for engine development. Differential equations of mass and energy conservation representing thermal state equations must be solved mathematically [5]. This approach to study the engine and its processes during the development stage will reduce high test bench costs and simplifies the whole process of engine development in the initial stages. In this work the engine model is built and simulated using the simulation tool GT-Suite.

A model of the engine which under study was developed by Jian Zhu of engine CAE-Fluids department at Volvo Car corporation. This model was developed to simulate the gas exchange and performance of the engine, in this work this model is being used to yield the necessary boundary conditions that are imposed on the thermal model described in the later sections.

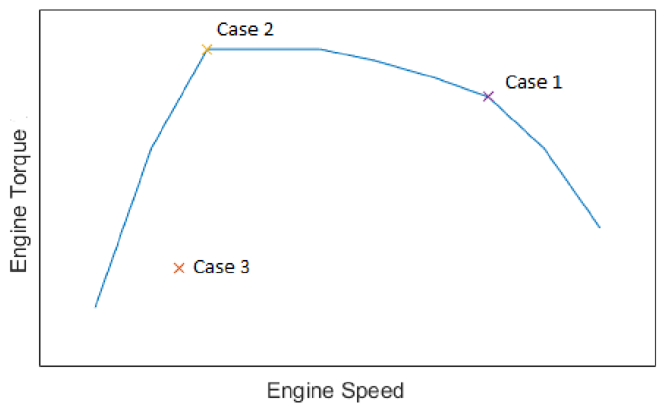

Figure 2.2: Engine Map and Load Points

The model was integrated with a thermal mass model and was simulated at three different operating points namely part load, full torque and full power operating conditions as shown in Figure 2.2 and the boundary conditions were fed to a thermal model for all operating points. Thus the integrated model allows to study the thermal behaviour of the engine at these operating points.

2.3 Engine Thermal Model

The combustion process involving the conversion of chemical energy in form of fuel to heat energy is the core of the thermodynamic cycle that results in production of mechanical power. An engine requires cooling, lubrication and exhaust after treatment to produce mechanical power effectively and efficiently. This whole process coupled with transmission of the mechanical power in an engine comprises of various parts which are involved in the heat transfer process. Apart from the utilized heat energy the remaining is released into environment through different means Figure 2.3.

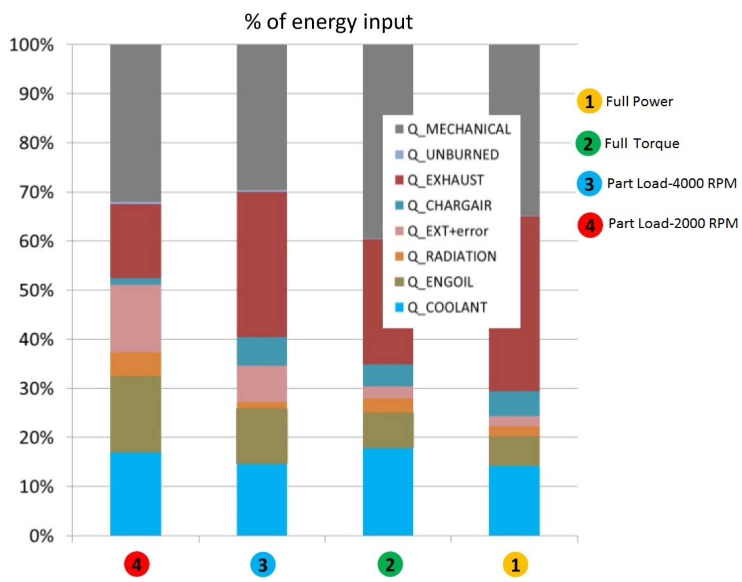

Figure 2.3: Heat rejection relative distribution as function of engine running conditions. [measurements of VED5 engine as measured at FKFS institute (Stuttgart, Germany) on Volvo Car Corporation commission] Courtesy of: Mirko Bovo, VCC.

Thus to perform a detailed study of the thermal behaviour of the engine a dedicated thermal model accounting for heat conduction, convection and radiation between different components of the engine is required. Considering all these, the problem will be a three dimensional, time dependent, multi-phase problem. The resulting model will be a high resolution three dimensional engine thermal model, one such model has been done by Bovo. M and Somhorst. J at Volvo Car Corporation[3].

The 3D model is accurate and offers high spacial resolution but the computational memory and cost of such model is very high. A fast running model can be used where accuracy is not a top priority, a fast running 1D thermal model in GT-Suite was built based on the high resolution 3D model as many application requires a fast running model though the accuracy is less. The advantage of a 1D model is that the complete steady state and transient thermal behaviour at different loads can be studied at comparatively lesser cost in the absence of the test rig or a expensive 3D model. Figure 2.4 describes the simulation domains and graphical representation of the 3D engine thermal model.

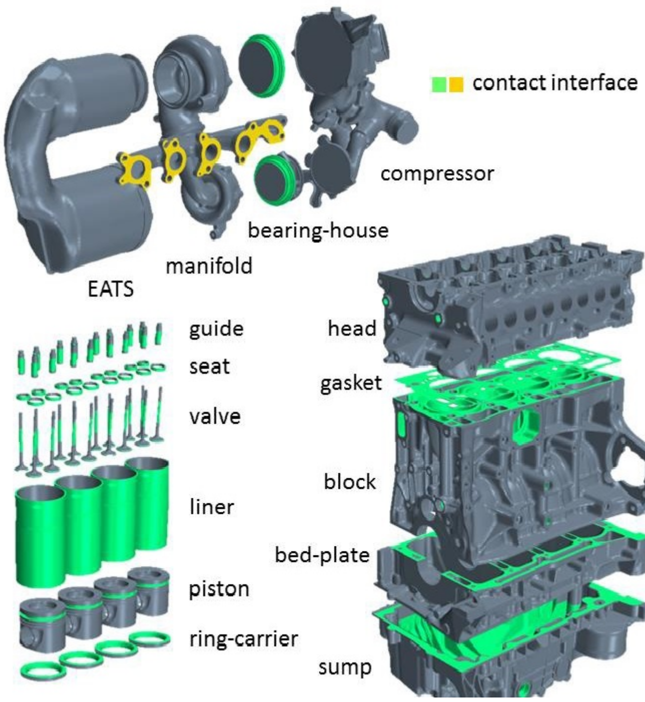

Figure 2.4: Graphical representation and simulation domains of the 3D model Courtesy of: Mirko Bovo, VCC.

A comparison study of the 3D thermal model and the 1D thermal model was done as a master thesis by Hesham Saad Eldin [4]. It was done by comparing simulation results of 1D engine thermal model in GT-Suite and a 3D engine thermal model done in Star-ccm+ V10.06.0009 for different thermal loads.

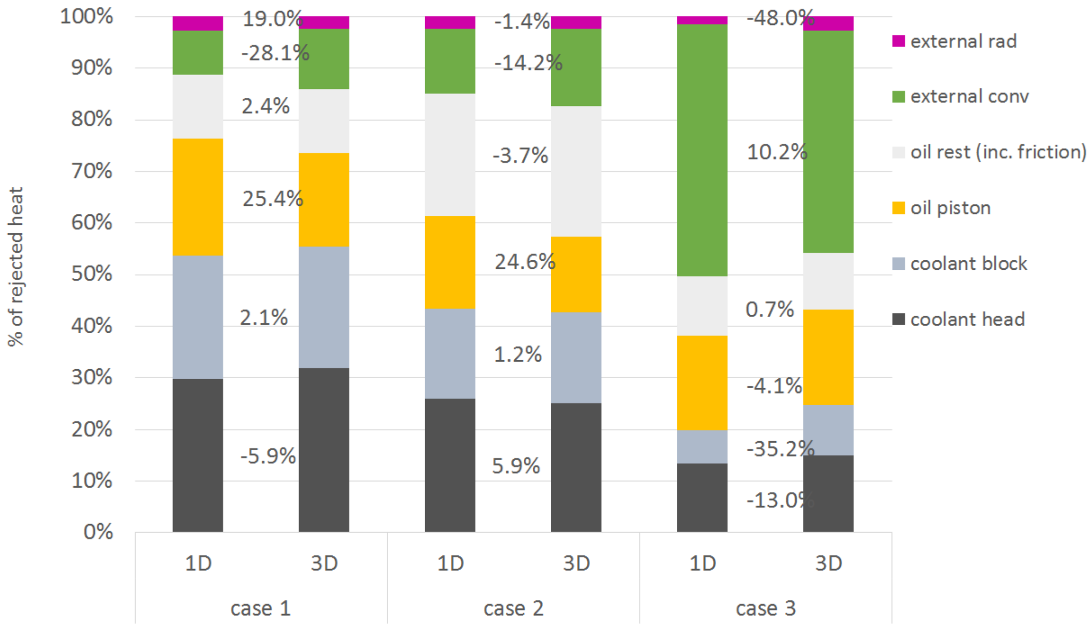

Figure 2.5: 3D vs 1D simulation result comparison Courtesy of: Mirko Bovo, VCC.

Figure 2.5 shows the 3D and 1D simulation results and they are in agreement with each other. This indicates that the 1D thermal model can be used to study the thermal behaviour of the engine effectively. The boundary conditions obtained from a 1D gas-exchange model simulated at different load points were imposed on the thermal models, In this approach the gas-exchange model and the thermal mass model are independent and the necessary boundary conditions were imposed independently. The friction and the wall temperatures imposed on the gas-exchange model were predetermined and similarly the boundary conditions obtained from the gas-exchange model were independent of the changes in the thermal model.

The one way heat load coupling method used in these models is a coarse approximation of reality as the heat load supplied by combustion is not constant but a function of wall temperature of the gas side surface.

Thus in this work a two way heat load coupling which accounts for dynamic exchange of boundary conditions is used. Thus the performance model and the thermal model are simulated together and the dependencies and the behaviour of each model can be studied in transient and steady state operating conditions.

3
Methods

The two models, performance and thermal models are to be coupled. Three different steady state operating conditions, part load, full power and full torque are considered. Furthermore a transient operating condition is considered to simulate a simplified engine warm up process. There are four different models used in this work. They are gas exchange model for part load and full load and the thermal model for steady state and transient operating conditions. The gas exchange model is used to simulate the combustion process at different loads, the boundary conditions obtained from this model is imposed on the thermal model. There is a physical interface between the gas and the solid. In the integrated model, this interface is represented by the connections between performance and thermal model. Apart from the operating points, major difference between the part load and full load gas exchange models is the approach used to model friction. In the part load model a provided map based friction calculation is used. This friction map was obtained from test rig measurements. In case of the full load model, the provided Schwarzmeier-Raulein friction model [7] is used. This model uses engine load (bmep), speed (rpm), oil and coolant temperatures to calculate the friction mean effective pressure.

In the transient thermal model, heat rejected is a function of time and this model is used to study the warm-up phase of the engine. In the steady state model, the heat rejected remains constant at all time intervals for a given load point. This model is used to study the temperature and heat transfer coefficients of different components of the engine and to calibrate the models based on the 3D results.

The thermal model used for transient operating conditions is not calibrated at full load operating condition hence it is not being considered in this work. Therefore the different integrated models as a outcome of this project are as described below.

- Steady state - Full Load model
- Steady state - Part load model
- Transient - Part load model

Prior to this project the gas exchange and the thermal models were simulated independently. The boundary conditions obtained from the gas exchange model were fed to the thermal model. The inputs to the thermal model from the gas exchange model are the gas temperatures and the heat transfer coefficient of different surfaces exposed to the thermodynamic cycle. The gas side surfaces of liners, piston, heads, valves, exhaust ports and manifold acts as a heat source to both the models. The heat generated by the combustion process is not constant and it is a function of gas side surface wall temperature. As the boundary conditions were obtained by simulating the models independently there is no instantaneous exchange of boundary conditions. Hence the transient operating condition cannot be simulated. Therefore, to simulate the transient operating condition both the models should be coupled to allow dynamic exchange of boundary conditions between the models.

In the following sections, the coupling approach, modelling description, details of parametric study, results and comparison are presented. The models built in this work are steady state full load (SSFL), steady state part load (SSPL) and transient part load (TCPL). They will be referred to with their respective names in upcoming sections. All the models are built in a simulation platform GT-Suite v2016.

3.1 Description of coupling approach

The thermal and the gas exchange model can be coupled in two ways in GT-Suite. They are indirect and direct coupling.

3.1.1 Indirect coupling

Indirect coupling is also called as partial integration because the boundary condition from only the gas exchange model is imposed to the thermal model. The gas exchange and the thermal models run separately. The boundary condition data from the gas exchange model is stored in the database and used during simulation of thermal model and there is no information transfer from thermal model to the gas exchange model. The indirect coupling method can be used only for steady engine speed and load or for transient speed and load in the absence of turbocharging, charge air cooler or EGR cooling. This type of coupling is used when the interaction between the gas exchange and the thermal model is not critical.

3.1.2 Direct coupling

In this approach, the gas exchange and the thermal model are merged and solved simultaneously. The boundary conditions from the gas exchange model are imposed on the thermal model and there is feedback from the thermal model. This allows the thermal model to get the gas boundary conditions and heat transfer coefficients from the gas exchange model and the gas exchange model to receive instantaneous wall temperatures from the thermal model.

The direct coupling method provides accurate and complete integration between the thermal model and the gas exchange models allowing dynamic exchange of information. This allows the study of thermal behaviour of the engine and the influence of changes in thermal model on the performance of the engine. The disadvantage of the direct coupling method is that the simulation duration is significantly higher compared to indirect coupling. In this project, direct coupling method is used as there is a need for dynamic exchange of boundary conditions and to use the model to study the engine behaviour in transient operating condition like warm up of the engine. Figure 3.1 represents indirect and direct coupling approaches.

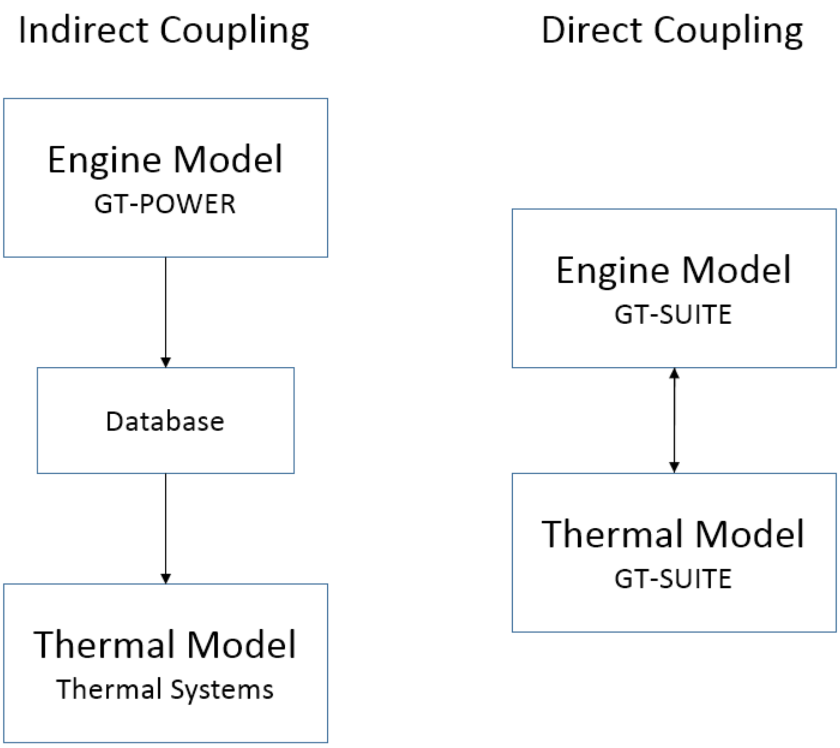

Figure 3.1: Comparison of different coupling methods Courtesy: Tutorial on advanced integration of engine and cooling systems. Gt-Suite 2016.

3.2 Modelling description

The gas exchange model can simulate only the performance parameters of the engine. The details of the heat rejected cannot be tracked accurately from the gas exchange model. By connecting the thermal model to the gas exchange model the thermal model the rejected heat can be tracked accurately. The integrated model also allows to study the effect of thermal properties on the performance of the engine. The friction in an engine is a function of oil and coolant temperature, the oil viscosity changes depending on the temperature and it influences the engine friction. As the engine fuel consumption is dependent on friction, the study of the effect of oil and coolant temperatures plays a important role in reducing the fuel consumption of the engine. The wall temperatures of the components in contact with the hot combustion gases influences the heat transfer and the gas boundary conditions. These conditions are very dynamic and it is important to have instantaneous data transfer between the gas exchange and the thermal models.

In the integrated model the components of the gas exchange model that are in contact with the combustion gases are connected to thermal masses having the same properties. The surface areas of few components are not exactly the same, the heat transfer rate is directly proportional to the surface area. To overcome this and to have a accurate track of heat transfer the surface area in the thermal mass part is considered for the calculation in all the integrated models. Figure 3.2 represents the different parts of the integrated model and the boundary conditions exchange between them.

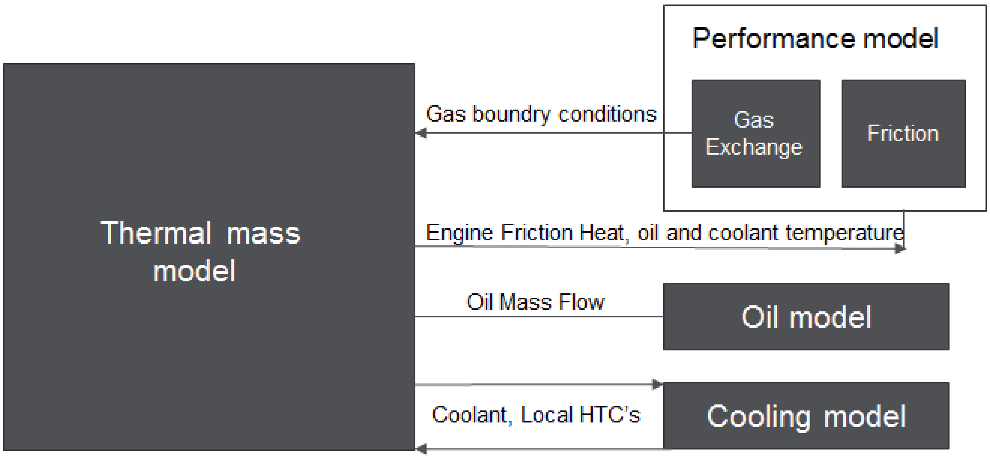

Figure 3.2: Representation of different parts in thermal model and boundary conditions transferred between them.

The gas exchange model and the thermal mass model are combined into a single project map in GT-Suite. The thermal masses representing the components in contact with the combustion gases are identified in the thermal folder of the integrated model, the surface areas of the components are matched and are linked. The connections transferring multiple boundary condition are linked through a "multiple thermal BConn" and the connections transferring heat transfer coefficients are linked through "convection conn" or "radiation conn" depending on the type of heat transfer. The initial state object in the thermal folder of the gas exchange part for all the components was set to wall temperature defined by FE structure. This step is used to calculate the wall temperature from the corresponding thermal parts. The gas boundary condition from the cylinder component in the gas exchange is imposed to the thermal model by selecting the engine cylinder object in the boundary condition folder. Thus the boundary conditions are exchanged without physical connection between the components in case of engine cylinder.

The turbines and the compressors are modelled as simple orifices and thus they do not have surface areas in the gas exchange part. Therefore the heat rejected from these components are dumped on to the respective thermal masses and the surface area of the thermal mass will be considered in calculations. The heat generated by friction is considered to be carried away completely by the oil. The friction heat quantity is extracted from the crank-train object in the gas exchange part and imposed on the oil heat source object in the thermal part through a dummy thermal mass. All the three integrated models are modelled with the above considerations. As discussed earlier the major difference between the three models are the operating conditions and the friction models used.

The oil and coolant temperatures plays a major role during engine warm up phase as the temperatures will be low and this increases the friction. Therefore it is important to study the effect of the oil and coolant on friction during warm up. Different strategies were used to warm up the oil and its effect on friction is studied using the transient integrated model. Different initial coolant and oil temperatures are considered and the difference in friction for all the cases are compared. The model validation is done by comparing the results with other base models namely 1D standalone and 3d models. Conservation of energy is verified for the steady state case.

The steady state models are used to study the temperature and heat transfer coefficients of different components, to verify energy conservation in models and also to compare the results to the standalone models. The comparison provides us with an insight into the effect that dynamic boundary conditions closer to reality have on the distribution of energy. Any change in overall fuel consumption can easily be traced for the same load conditions. Changes in energy being rejected in various components and the exhaust will also be noted.

The transient model is used to study warm up conditions and the effects of various strategies on warm up of different components. The strategies investigated in this thesis work are:

- Engine oil cooler coupled - EOC coupled from the engine startup
- Engine oil cooler decoupled - EOC decoupled from the engine startup
- Engine oil cooler decoupled after warm up - EOC coupled at startup and decoupled after warm up

- Oil Volume 6 Litres
- Oil Volume 3 Litres

It is also used to conduct a parametric study of the dependence of friction on oil and coolant temperatures individually. This is not a physically representative test but is done to study the behaviour with respect to one variable.

- Coolant temperature varying, Oil temperatures - 20, 50 and 90 C
- Oil temperature varying, Coolant temperatures - 20, 50 and 90 C

The results of steady state models, transient model and the parametric study are presented in next section.

4

Results

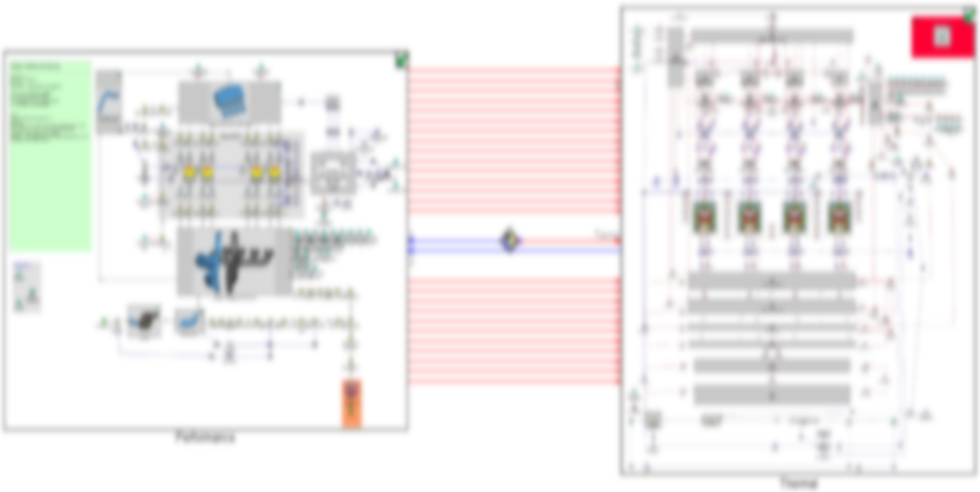

Figure 4.1: Image of the final configuration of the completed model. It has been blurred out as the engine is still in development

Figure 4.1 shows the complete integrated model of the engine. The block on the left side shows the gas exchange model. It is the driving part of the model. The block on the right side is the thermal model. The 4 red blocks seen are the reactors or cylinders. They are directly linked to the cylinder objects in the gas exchange model. It is a template available in GT-Suite. The data about the geometry of the cylinder etc are imported from the gas exchange model. The thermal link between them provides dynamically changing gas boundary conditions. The flow circuits for oil and coolant outside the block are modelled using pipe objects.

All the parts in the thermal model are connected to their corresponding counterparts in the gas exchange model. There are 29 external (red lines) and 20 internal links between the two models. The two blue lines indicate the oil and coolant temperature being input to the friction model in the gas exchange model. The internal links are in the properties of the object to thermally link them to the corresponding parts in the other model.

The gas exchange model run in the crank angle degree domain and the thermal model runs in the time domain. This required that they be run with different time control flags. The gas exchange model can only be run in the periodic mode. The thermal model, being in the time domain, is run in continuous mode. The step size for gas exchange model is 15 CAD and for the thermal model is 0.1s.

4.1 Steady state results

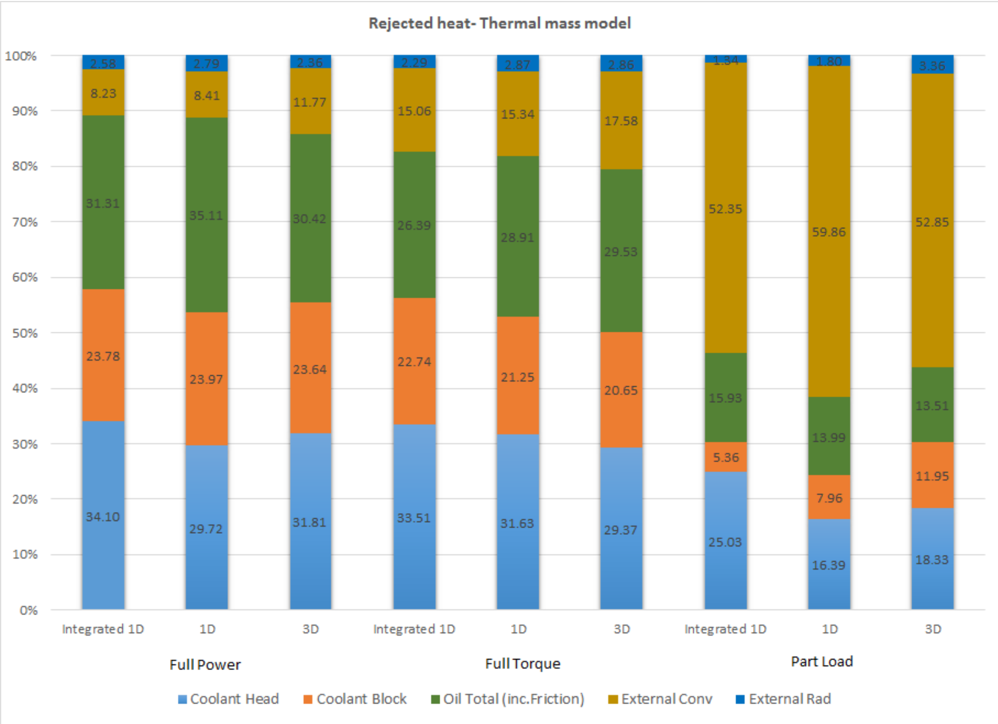

Figure 4.2: Rejected Heat from the Thermal Model

The steady state model is configured for 3 load cases in 2 models. The consistency of the models is verified by applying the energy conservation principle. The comparison of heat rejected to different components as predicted by the stand alone models and the integrated models was done. The effect of coupling the models together on fuel consumption was also done.

Figure 4.2 shows the distribution of heat rejected for different load cases. It can be seen that, in general, there is an increase in the ratio of exhaust heat in the integrated models when compared to the standalone models. This can be attributed to the fact that the standalone models have fixed heat transfer coefficients between gases and surfaces, whereas the integrated model has dynamically changing heat transfer coefficients. This may have led to less heat being rejected through the coolant, oil and hence more heat being rejected through exhaust.

4.1.1 Full Power

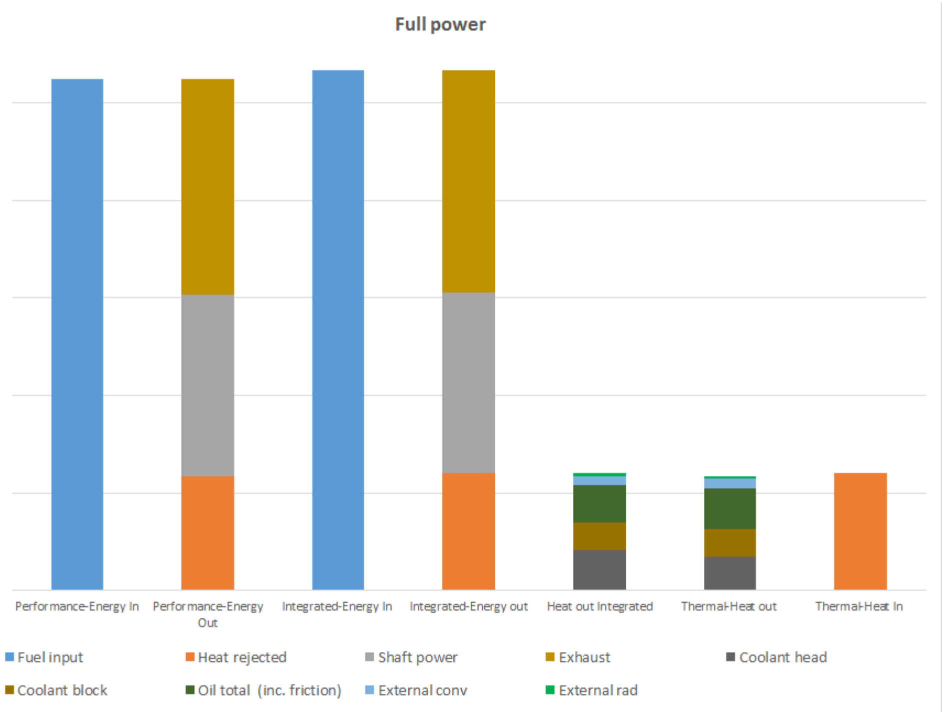

Figure 4.3: Energy Balance for SSFL model - Full Power.

The Figure 4.3 summarizes the results of the SSFL model in the full power case. The "Performance Energy in and out" bars are the results from the gas exchange standalone model. The sum of heat going out is a match to the fuel energy coming in. The heat rejected forms the input to the thermal model which is seen on the right side - "Thermal Energy in and out". The thermal model predicts the amount of heat being taken away through different sinks.

The integrated model is the combination of the two models with dynamic exchange of boundary conditions. It is clearly seen that this causes a change in how the energy is distributed. It is to be noted that the total fuel energy going in is same as the total energy coming out, and that the heat rejected from the gas exchange model being fed into the thermal model as input is the same as the total output from the thermal model. Thus verifying energy conservation. It can also been that after the integration, the prediction of energy lost in exhaust has increased and less heat is predicted to be rejected through other sinks.

4.1.2 Full Torque and Part Load

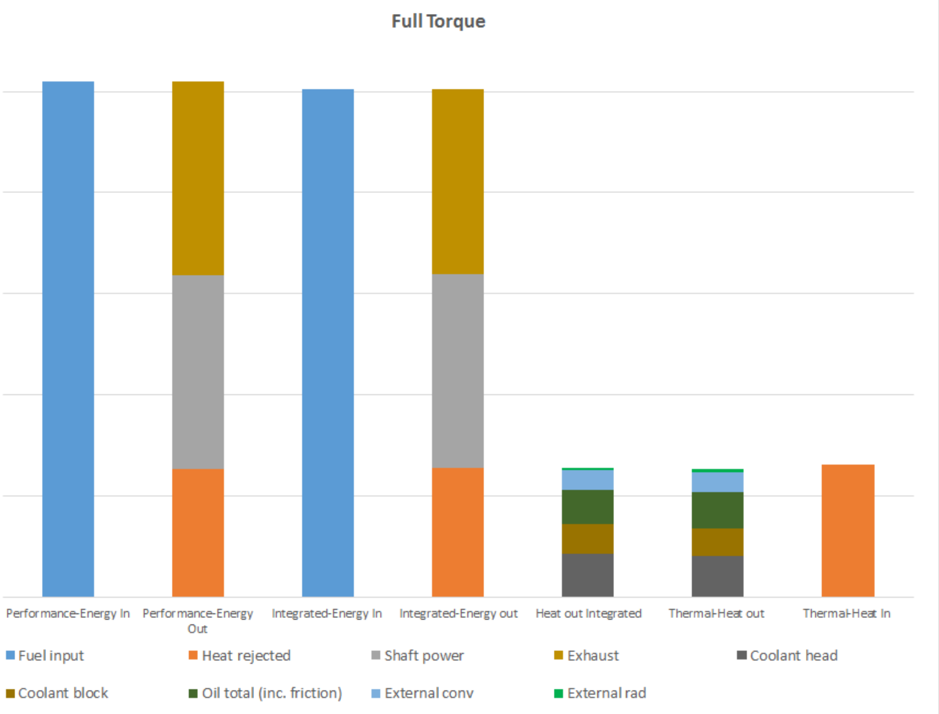

Figure 4.4: Energy Balance for SSFL model - Full Torue.

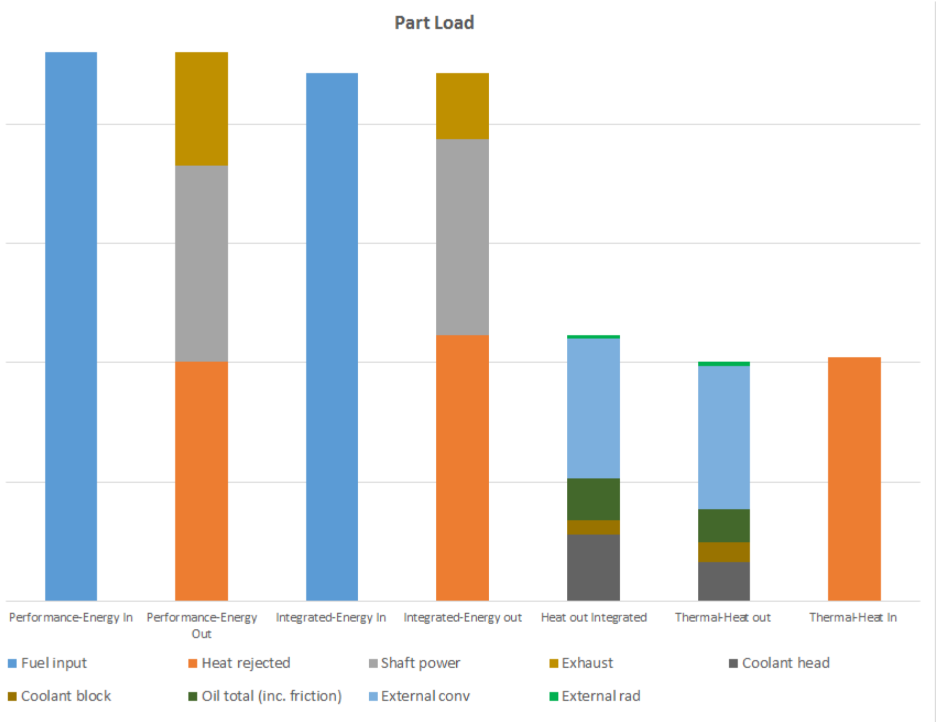

Figure 4.5: Energy Balance for SSPL model.

The Figure 4.5 summarizes the results of the SSPL model. Here, we see a decrease in exhaust energy and an increase in heat rejected to coolant, oil and environment. This can be attributed to more accurate calculation of temperatures and the fact that they are low at part loads. We can also see an overall reduction in fuel input for the same load in the integrated model.

As seen before, there is close a match between the fuel input and the total energy output in the model. The heat rejected also matches the heat processed by the coolant oil and the environment. Hence the model is energy conservative.

4.2 Transient results

The simulations for transient conditions were done by setting the initial temperatures of all the components to 20 C. The warm up of various components was hence captured.

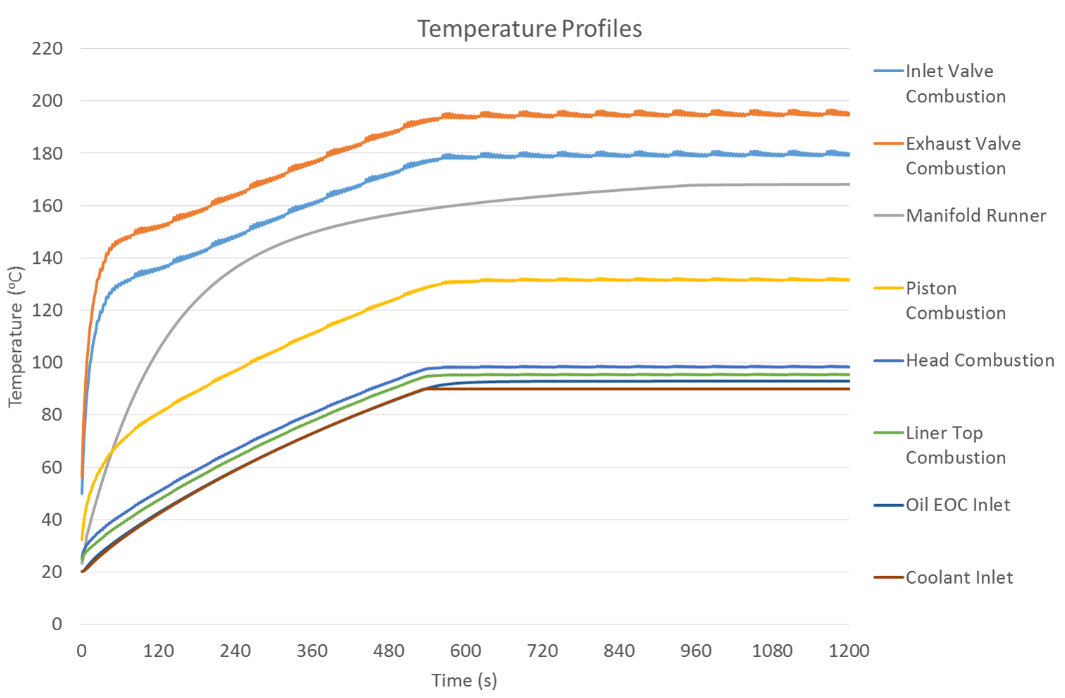

Figure 4.6: Temperature profile of various components in the transient model.

Figure 4.6 shows the warm up of various components of the engine. These results for different strategies can be used to study the effect the strategy has on the warm up on any component of interest in the engine. For this example, it is seen that at part load, head and liner follow the same temperature profile as that of the oil. The amount of energy being processed at part loads is quite small when compared to the amount of energy the cooling system is designed to handle. Thus, the temperature profiles of the parts in contact with oil or coolant tend to follow similar temperature profile. As expected the temperature at the valves is the highest. But this may not necessarily be the case at higher loads.

A Schwarzeier-Raulein friction model [7] is used. This model calculates the FMEP based on oil and coolant temperatures. These temperatures are taken from the thermal mass model and hence the variation of friction as a function of warm up can be studied. The engine has to overcome friction torque and deliver the required brake torque and thus has a influence on the fuel consumption.

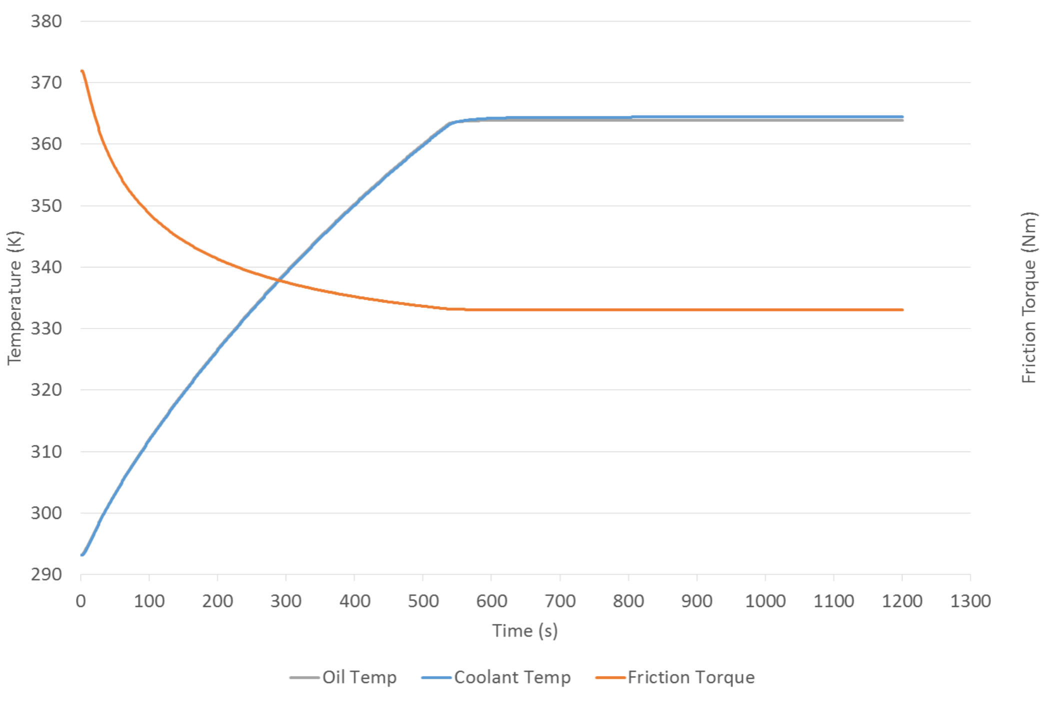

Figure 4.7: Oil and coolant temperatures and friction torque vs time for EOC Coupled.

The Figure 4.7 shows the temperature profile of oil and coolant and the variation of friction torque as the engine warms up. We can see that friction torque is about twice as high for a cold engine as that for a warm engine. This is in agreement with strip down measurements of various engines and results form other studies.

Following the completion of the model, different strategies of warm up were investigated.

4.2.1 Engine Oil Cooler (EOC)

The engine oil cooler is designed and sized for operation at full load conditions. Hence at part load it can be assumed to be working perfectly which practically results in the temperature of the oil outlet is equal to that of the coolant inlet.

It is interesting to see how decoupling the EOC under warm up conditions affects the temperature profiles of the oil and coolant and therefore friction.

4.2.1.1 EOC Active

The Figure 4.7 shown above is with the EOC active. It is seen that the oil and coolant follow the same profile as the EOC is effective ideal at part load conditions.

4.2.1.2 EOC Bypassed

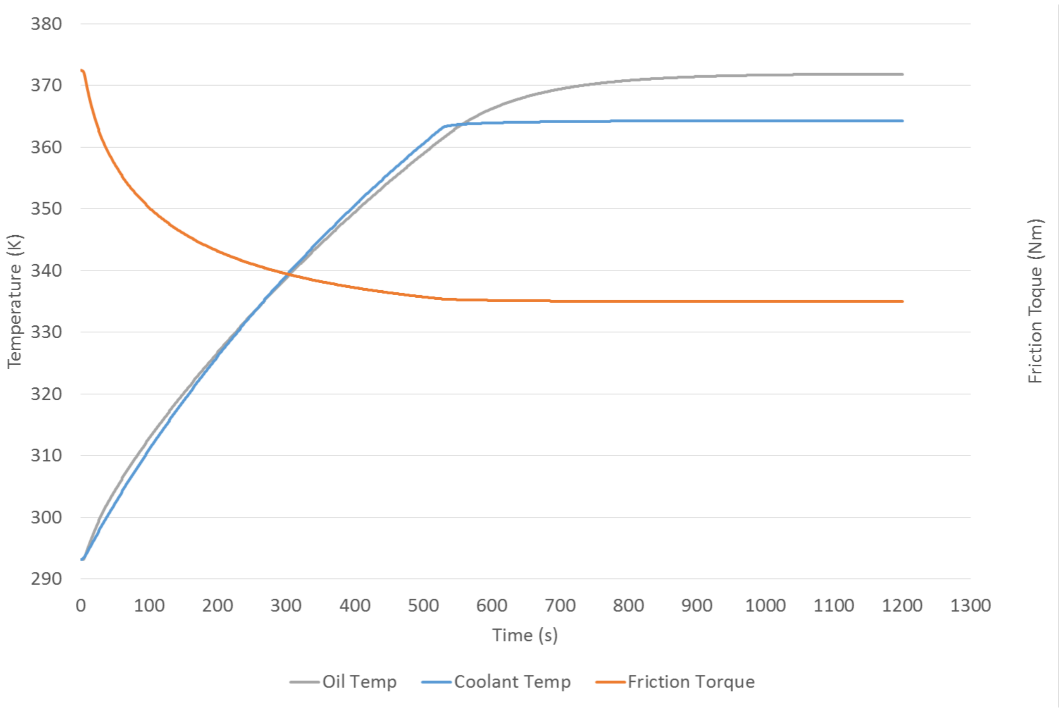

Figure 4.8: Oil and coolant temperatures and friction torque vs time for EOC Bypassed.

The Figure 4.8 shows the oil and coolant temperature profiles and the friction torque as a function of time. It is clearly seen that the oil heats up quickly at the beginning. It is then very close to the coolant temperature. After the thermostat is trigger in the coolant circuit, we can see that the oil continues to warm up to about 10°C more than the coolant. This results in a small decrease in final friction torque. But the friction profile is almost the same with very small differences.

Even though the EOC is decoupled in this case, the amount of interaction between the oil and coolant in the engine block seems to keep the temperatures of the two pretty close. It is also important to note that the coolant heats up faster after a certain amount of time. While the higher steady temperature of oil is good for friction, it being lower than coolant shows that it might be better to couple the EOC during the warm up phase.

4.2.1.3 EOC Decoupled after warm up

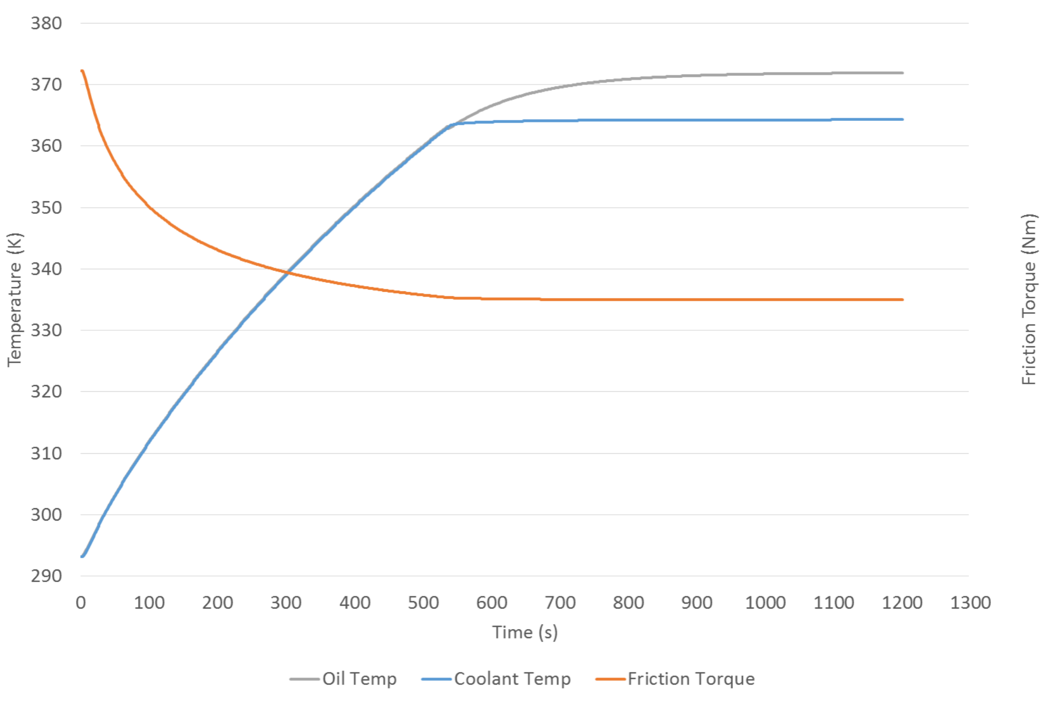

Figure 4.9: Oil and coolant temperatures and friction torque vs time for EOC Decoupled after warm up.

It can be seen in the Figure 4.8 that the oil temperature that of the coolant. This is in general undesirable and so it would be interesting to see the results when the EOC was bypassed only after the warmup stage.

The Figure 4.9 shows the oil and coolant temperature curves and the friction as a function time. We can see that the oil and coolant temperatures follow the same profile till warm up after which the EOC is decoupled and the oil continues to warm up till about 10°C higher than the coolant.

It can be seen that any differences in the fuel consumption figures, while being conceptually verified, are not practically significant. But these have to be kept in mind when combining with other strategies which can compound the effects. For example, if the volume of oil available for warm up was halved, the oil might warm up even fast. To investigate this, the volume of oil is changed and the effects are studied.

4.2.2 Oil Volume 6 Litres VS. 3 Litres

As mentioned above, changing the oil volume might compound the effects on fuel consumption when combined with couple/decoupling the EOC. Lower volume of oil warms up faster and hence lowers friction. This should result in reduced fuel consumption.

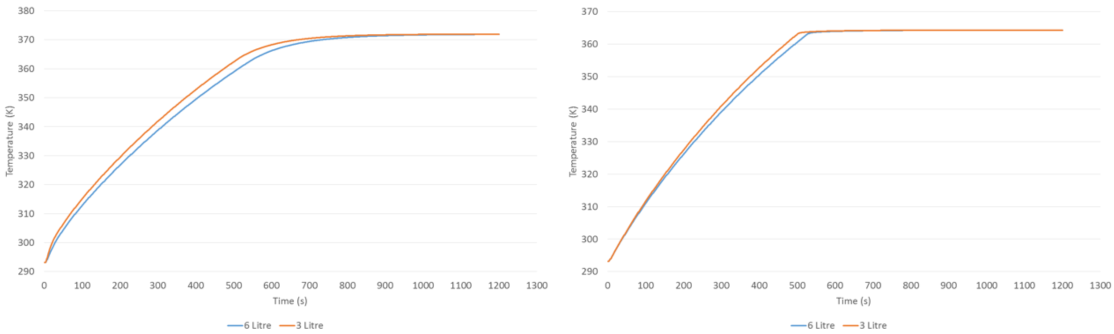

Figure 4.10: Oil and coolant temperature for 6l and 3l oil volumes.

Figure 4.10 shows the warm up of oil and coolant in the EOC decoupled setting. It is clearly seen that the oil and coolant warm up faster when the oil volume is reduced. This is quite intuitive as the lesser quantity takes lesser energy to war up. As for its effect on the fuel consumption, the results are tabulated as follows in Table 4.1.

Table 4.1: Fuel Consumption for various configurations

EOC Setting	FC for Oil 6l (%)	FC for Oil 3l (%)
Coupled	–	0.08
Decoupled	0.01	0.14
Decoupled after warm up	0.05	0.13

It is clear that there is a decrease in fuel consumption for the respective cases when oil volume is reduced from 6 litres to 3 litres. The reduction of fuel consumption is from the warm up phase only as it is clear that the saturation temperature of the oil and coolant in either case remains the same. It is therefore safe to say that this strategy can potentially be used on conjunction with other strategies for better warm up.

4.2.3 Parametric Study of Friction

As can be seen in the EOC decoupled configuration, the oil temperature and coolant temperature increase at varying rates at different phases of warm up. To better understand the dependence of friction on these two parameters individually, a parametric study is conducted. This is done with a model configuration which is not physically representative but will give an overview of the friction behaviour as a function of either temperatures.

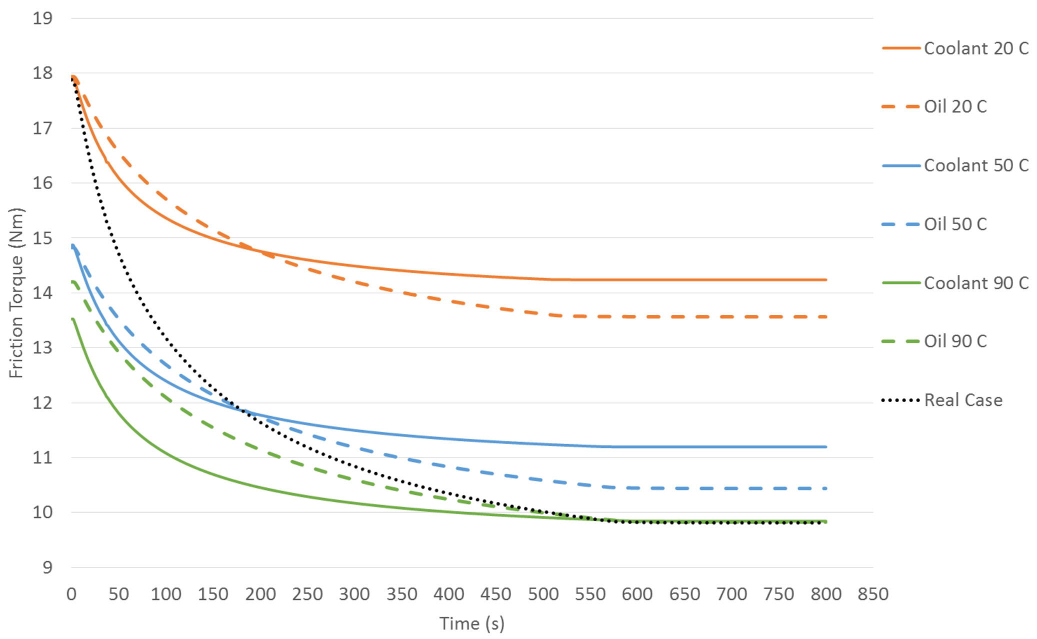

Figure 4.11: Parametric study of friction as a function of oil and coolant temperatures.

The Figure 4.11 shows the variation of friction with varying oil or coolant temperature with the other one being constant. It is clear to see that when either one is held at 20°C the friction is highest over all. But it is interesting to note that the final friction is lower when the oil is cold than when the coolant is cold. It should also be noted that in the initial warm up phase, friction drops faster as oil warms up than as the coolant warms up. This trend is seen even when either temperature is held at 50°C. In the friction curves of oil and coolant at 90°C, it is seen that friction is lower for cold coolant than cold oil.

What can be taken away from this study is that in the initial warm up phase it is beneficial to warm up the oil faster, and later to warm up the coolant. This study only gives us an indication of how friction responds to oil and coolant temperatures individually. It is not physics based model.

5

Conclusion

From the results of the steady state model, we can conclude that the model is verified for full power and part load conditions. The model is not calibrated or setup properly for full torque case. It is seen that in the full torque case the shaft power is not matching the required output. Regardless, having reasonable results for the part load steady state model, the transient model could be developed.

The initial results of the transient model are qualitatively plausible. This allowed investigation of different strategies for warm up phase. It is seen that there is change in fuel consumption, albeit very small, and energy distribution. Reducing oil volume to 3l reduces the time for warm up and reduces friction faster. This results in lower fuel consumption. The decoupling of EOC and it's timing also has an effect on the friction behaviour.

The friction model used in this model is not calibrated for part load conditions. The friction behaviour was qualitative as expected but the absolute value was very high. It was altered to match the friction of a warm engine according to the friction map provided. But it is clear that this value is critical to the warm up profiles of oil and coolant.

The parametric test also showed that in the initial stages of warm up, the oil warm temperature has a bigger influence on friction then coolant temperature. This can be attributed to the high viscosity the oil has at low temperatures. At higher temperatures however, the coolant temperature has a bigger influence on friction. This might be because of the thermal expansion of the surfaces in contact with the coolant.

The results of this model alone are limited to studying changes in fuel consumption and energy distribution. But it's value is not limited to this. There are a number customers for the energy being distributed. They all have different requirements with some having a higher priority than others. This model will allow testing of different strategies which will help in addressing the demands of these customers. Thus there will be changes in fuel consumption in many different indirect ways.

For example, the exhaust gas after treatments team need information about the amount of heat in the exhaust. They are an important customer of this heat as the EATS needs to be warmed up as quickly as possible. Another important customer for rejected heat is cabin HVAC team. In certain conditions, the foremost priority is to heat up the cabin and it is a challenge to provide the said heat. This model can be used to simulate a lot of different conditions, strategies and study the output of choice.

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